

CFD ANALYSIS OF MATRIX COOLING METHOD IN GAS TURBINE BLADES

*A thesis submitted in partial fulfilment of the requirements
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Engineering By

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CERTIFICATE

This is to certify that the work in this thesis entitled “CFD analysis of matrix cooling method in gas turbine blades using fluent” by ***Akhilesh Behera***, has been carried out under my supervision in partial fulfillment of the requirements for the degree of *Bachelor of Technology in Mechanical Engineering* during session 2014-2015 in the *Department of Mechanical Engineering, National Institute of Technology, Rourkela*.

To the best of my knowledge, this work has not been submitted to any other University/Institute for the award of any degree or diploma.

Date: 29/05/2015

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ABSTRACT

Gas turbines are extensively in use for aircraft propulsion, land-based power generations, and various industrial applications. Thermal efficiency and the power output of a gas turbine increases with increase in turbine rotor inlet temperature (RIT). The current RIT level in many advanced gas turbines is far above the melting point of the used blade material. Therefore, along with development in high temperature material, a more sophisticated cooling scheme must be developed for continuing the safe operation of gas turbines with high performances. Gas turbine blades can be cooled internally as well as externally. This paper is focused on the internal cooling of turbine blades and vanes of a gas turbine. Internal cooling can be achieved by passing coolant through various enhanced serpentine passages inside the blade and extracting heat from outside of the blades. Jet impingement, matrix cooling, rib turbulator, dimple and pin fin cooling are utilized as the methods of internal cooling, which are presented in various articles. Due to the different enhancement in heat transfer and in pressure drop, they are being used in specific part of the blades and the vanes on a gas turbine. The matrix cooling, also known as lattice-work or vortex cooling provides a good strength to blades by the layers of ribs which intersect each other from the opposite wall. A significant increase in the heat transfer is obtained due to an increase in heat transfer area, impinging and in swirling flows (which helps to promote turbulence), induced by the geometry of the matrix cooling channels.

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1. INTRODUCTION

1.1 Gas Turbine Basics[16]

There are three main parts in a gas turbine, namely: the compressor, the combustor and the turbine, as represented in Figure 1 by the numbers (1), (2) and (3) respectively. The function of the compressor is to compress the air before it goes into the combustion chamber, where it is mixed with the fuel and is ignited. This fuel-air mixture burns at high temperatures and expands. Thereafter the hot gas enters into the turbine and strikes the vanes, which direct the incoming gas to the blades. The blades are deflected by the oncoming gas stream and thus a torque is being produced on the shaft causing it to rotate which is then converted into useful work. One use of this rotational movement is to produce electricity by rotating a generator (4) and then stepping up by using a transformer (5).

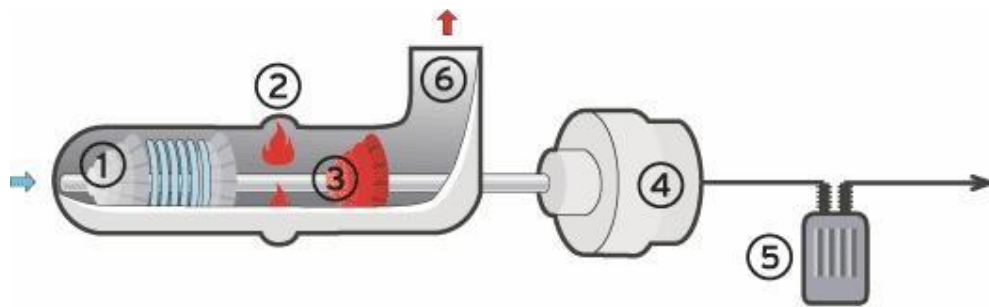


Figure 1:- Schematic of Gas Turbine

1.2 Limitations on Turbine Inlet Temperature (TIT)[16]

Due to its working nature, the power that is generated by a gas turbine increases with an increase in the temperature at which gas enters, known as the “turbine inlet temperature”. An increase in power output results in a higher efficiency. However, the turbine inlet temperature cannot be increased rapidly because of the limits imposed due to the temperature at which the blade material melts. Although some advances have been made in the material science to make some new alloys having high melting points that can withstand the operation at such a high temperatures without even failing, these materials are very expensive and are very difficult to machine.

1.3 Need for cooling[16]

As the blade material melts at a lower temperature than the operating conditions of the turbine, a cooling method must be incorporated into the blade design to ensure the safe and smooth running of the turbine. It is important, while devising a cooling scheme, to have knowledge about the boundary conditions of the blade during turbine operation, so that large gradients can be avoided. This is because large gradients cause thermal stress cutting the component life short significantly.

1.4 Turbine Cooling Basics[16]

Although cooling is necessary, it affects the gas turbine operation inadvertently:

1. The cooling air supplied to the blades and vanes is directly bled from the compressor.

As a result the mass of air going into the combustor is decreased.

2. In order to incorporate the various structures like fins, cooling passages etc. the trailing edge thickness of the blades must be increased which adversely affects the aerodynamic performance of the blades

Various parts of the turbine blade are cooled using various techniques. The front part, called the leading edge, is generally cooled by impingement cooling. The middle part is generally cooled by using snake-like passages endowed with ribs along with local film cooling. The back part, called the trailing edge, is generally cooled by impingement and film cooling.

1.4.1 Types of Cooling

There are two broad categories of cooling used in gas turbine blades:

- (a). Internal Cooling
- (b). External Cooling

In internal cooling, the cool compressed air flows internally within the passages of the turbine blade and thus heat transfer occurs between the cold air in the passage and the adjacent hot surface of the blade.

In external cooling, the cool compressed air is ejected from holes on the surface of the blade or the vane and creates a thin film between the surroundings and the blade surface thus preventing contact between the hot air and the blade surface, enhancing heat transfer.

1.4.2 Types of Internal Cooling

There are various types of internal cooling which have been developed over the years. No particular type of cooling is suitable for all blades for all applications. Thus the cooling scheme must be selected according to operating conditions and requirements of the application at hand.

1.4.2.1 Impingement Cooling

It is generally used near the leading edge of the airfoils where the jet of cooling air strikes the inside of the blade surface and hence the name impingement cooling. This technique can also be used in the middle part of the blade. The heat transfer characteristics of this kind of cooling depends on the size and distribution of jet holes, cross-section of the cooling channel and the surface area of the target face.[4]

1.4.2.2 Pin Fin Cooling

Since the trailing edge of the blade is very narrow, it is difficult to manufacture holes and passages in this portion, thus pin fin cooling is generally applied in this region. The flow around the pins is similar to flow around a cylinder. The air flow separates and the wakes are shed downstream. Moreover a horse shoe vortex also forms wrapping around the fins and creating additional mixing and thus enhancing heat transfer. The heat transfer characteristics largely depend on the type of fin array and the spacing of the pins in the array, the pin shape and size. [4]

1.4.2.3 Dimple Cooling

This type of cooling occurs due to the presence of concave depressions or indentations on the surfaces of the blade passage. They induce flow separation and reattachment and thus enhance heat transfer. They are a very desirable cooling technique as they have low pressure losses. [4]

1.4.2.4 Rib Turbulated Cooling

This type of cooling requires the usage of turbulence promoting structures on the walls of the cooling passage in the blades, which are cast along with the blade during manufacturing.

Heat is conducted through the blade wall is transferred to the coolant passing internally through the blade. The heat transfer characteristics largely depend on the aspect ratio of channel, rib configurations and Reynolds number of the coolant flow. [4]

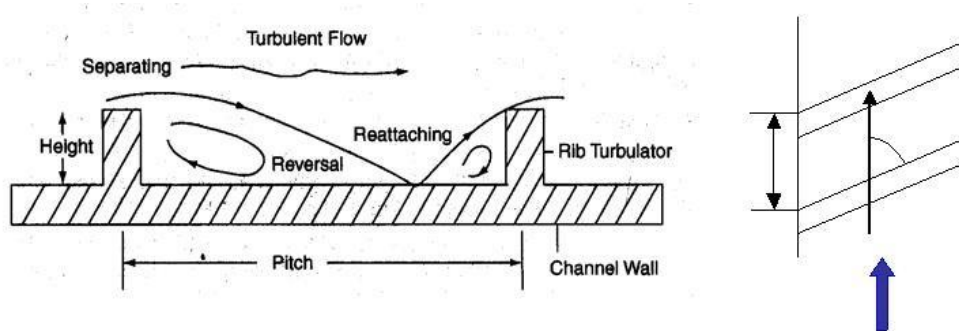


Fig 2 :- Mechanism of cooling in ribs

2. LITERATURE REVIEW

Bunker [2004][16], Nagoga[2000][16], Filipov and Bregman[2005][16] have worked on the analysis of the matrix cooling method of the gas turbine blades. The following is the brief summary of their work.

Bunker, 2004:-

He made some tests to study the variation in pressure losses and local as well as average Heat transfer coefficient in matrix cooling channels. He used two methods.

In first method, he used an Acrylic model and did not consider the effect of increased heat transfer area as the rib material was insulating. He then investigated the Heat transfer on the matrix shell corresponding to suction and pressure sides. He used liquid crystal techniques for temperature measurements.

In second method, he used metal model and did consider the increased heat transfer area. He used Infrared camera to measure temperature. He then compared the heat transfer in the matrix channels to that in a smooth channel calculated by Ditter-Boelter equation.

Nagoga, 2000:-

1. He worked on finding out the correlations for Heat transfer and friction factor from Heat transfer and flow studies.
2. The rotation of flow that is induced by the turn of flow at the side boundary of matrix intensifies the heat transfer and friction factor in the channel. The friction, heat transfer and rotation is maximum right after the spatial turn and decrease with increasing distance from the boundary wall. All these 3 factors depend on β , with maximum at $\beta = 45^\circ$.
3. The changes in Heat transfer is mainly along the perimeter of the concave surface and is maximum after the mid of the turn in the direction of flow. The rate of heat transfer is less than that of impingement on the concave surface.
4. After a lot of comparison between the matrix cooling method and other methods, he found out that matrix cooling methods are more effective than other methods concerning the cooling effectiveness.

5. After various numerical and practical tests were performed, he found out that the matrix helps to increase the life of the blade and vanes in high pressure turbines up to 40 times, which is 3-4 times higher than that of other methods.

Filipov and Bregman, 2005:-

He recommended that for an effective cooling, matrix should be used for airfoils with height to chord ratio ≤ 2 . For a greater height to chord ratio, the ratio of inlet to outlet area will be too small. It should be less than 2.

He spotted out some advantages of matrix cooling method-

- Increase in blade and vane strength.
- Very effective in blocking the internal cavities.
- High velocity outlet, even if there is some cases of internal cavities.

He also spotted out some disadvantages –

- Lower in average heat transfer enhancement factor than in pin fins and ribbed ducts in trailing edge area.
- Due to small thickness in slot, fin effectiveness is found out to be low at the trailing edge outlet.

3. METHODOLOGY

3.1- CAD Model:-

The blade profile was generated using SOLID WORKS 2012 software. The cross-section of blade was made by importing 374 points from a patent [1] into SOLID WORKS using MS-Excel 2007 and then a spline was drawn using those points. The spline was then extruded to a length of 150 mm.

A cooling channel was constructed as per the specifications used in [3]. The cross-section of the cooling channel was square with length 12 mm and breadth 12 mm. Matrix shaped ribs were made on top and bottom side of the cooling channel with a height {e} of 0.6 mm and spacing between ribs, pitch {p} of 7.488 mm approximately. 15 such ribs were made in matrix structure. The hydraulic diameter { D_h } of the channel was calculated to be 24 mm.

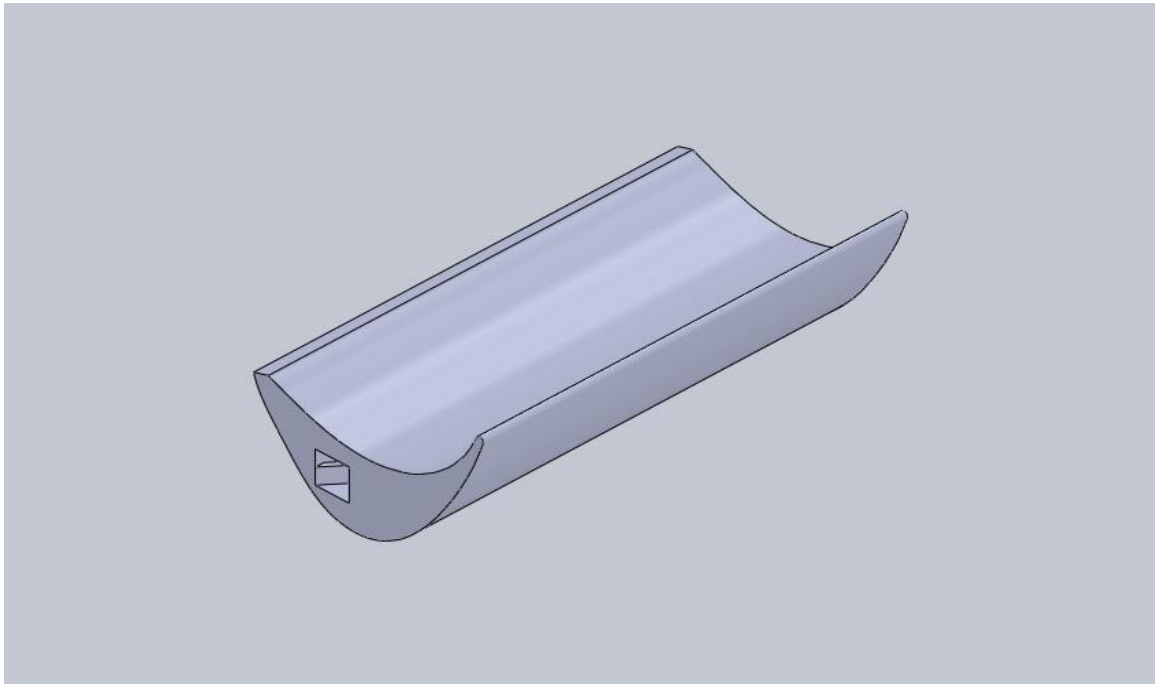


Fig 3 :- CAD model of the turbine blade in SOLID WORKS 2012.

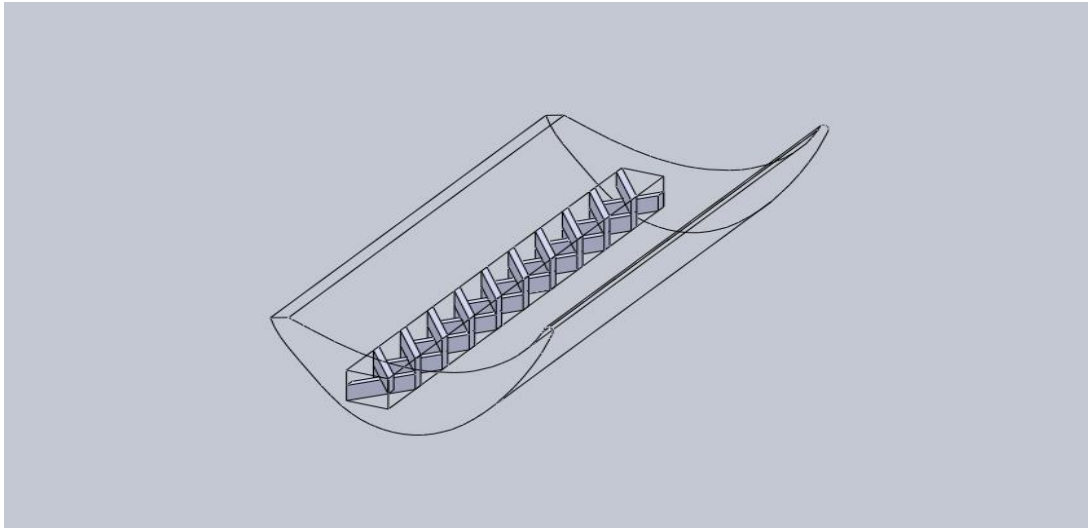


Fig 4 :- Wireframe structure showing inner matrix shaped ribs.

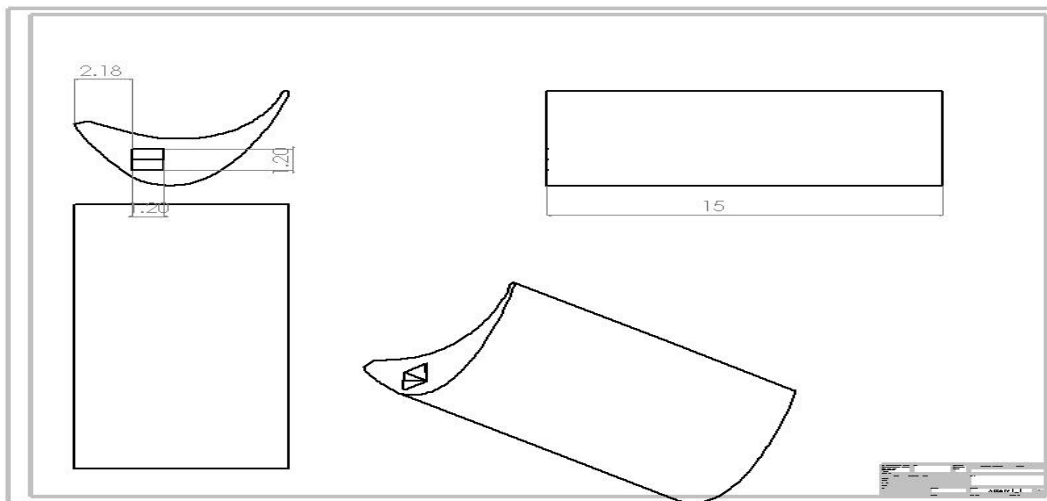


Fig 5 :- Scaling (in centimetre) of the model using SOLID WORKS 2012

3.2- Mesh:-

The CAD geometry was imported into ANSYS 15.0 Design Modeler and a fluid flow domain was constructed using the Design Modeler. Both the fluid flow domain and the blade were meshed using ANSYS 15.0 mechanical module. To obtain good results, a fine mesh was generated near the channel walls and the fluid flow domain so as to capture the velocity variations because of and the temperature variations. Other parts of the blade and fluid flow domain were meshed in such a way that good results could be obtained without being too computationally intensive.

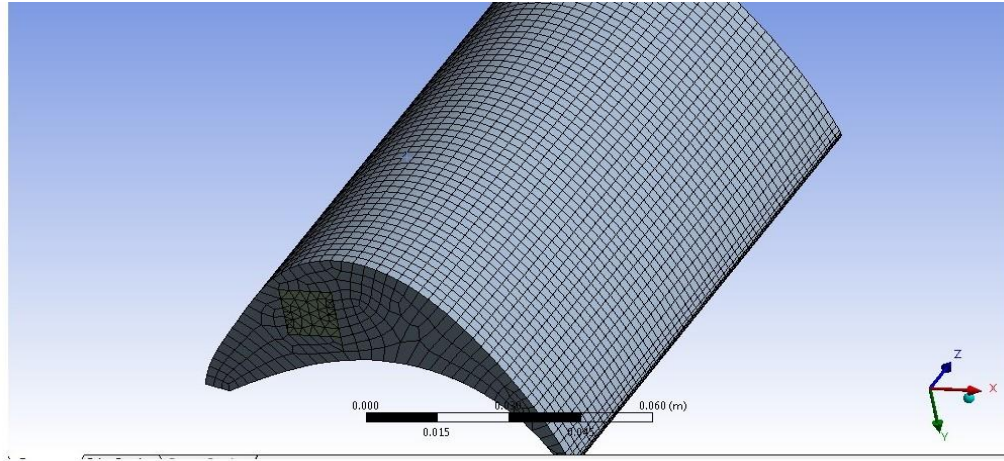


Fig 6:- Meshing using ANSYS 15.0

3.3- Set Up:-

The blade cooling problem was modeled as a Conjugate Heat Transfer {CHT} problem and with the use of ANSYS 15-FLUENT, it was setup and simulated. The fluid material was chose to be air from ANSYS 15-FLUENT database. The blade material was chose to be a nickel super-alloy Inconel 718, which is being used widely in aerospace related applications. The material properties were being imported from and are listed as below

Material Name	Thermal Conductivity {k}	Density {ρ}	Specific Heat Capacity {C _p }
Inconel 718	11.4 W/m-K	8190 kg/m ³	435 J/kg-K

3.3.1 Boundary conditions

The boundary conditions used were as specified in [3]. Five different mass transfer rates were being used for the ribbed matrix channel. For smooth channel, only one flow rate of $m=0.01$ kg/s was being used in order to compare with that of the ribbed matrix channel.

- Mass flow rates{m}:- 0.01 kg/s; 0.02 kg/s; 0.03 kg/s; 0.04 kg/s ; 0.05 kg/s
- Convective Heat Transfer Coefficient{h} of outer surface of blade :- $1000 \text{ W/m}^2\text{-K}$
- Free-stream temperature of the surroundings{ T_{free} }:- 1700 K(approx)
- Temperature of air at the inlet{ T_{inlet} }:- 400 K(approx)

3.3.2 Solution methods

Energy equation was turned on and the (k - ϵ) model, with standard wall functions was used to model the turbulent behavior.

- Scheme:- SIMPLEC
- Skewness Correction:- 1
- Gradients:- Least Square Cell Based
- Pressure:- Linear
- Momentum:- Power Law
- Turbulent Viscosity{ k):- Power Law
- Turbulent Dissipation { ϵ):- Power Law
- Energy:- Power Law

4. RESULTS:-

For $m = 0.01$ kg/s; $v = 50.93$ m/s, $m = 0.02$ kg/s; $v = 100.78$ m/s,

$m = 0.03$ kg/s; $v = 151.17$ m/s, $m = 0.04$ kg/s; $v = 201.56$ m/s and

$m = 0.05$ kg/s; $v = 251.95$ m/s:

the following velocity and temperature contours have been observed in order to analyse the variation of temperature and velocity along the blade's surface as well as the fluid domain's surface.

- Temperature contour on the side walls, inlet and outlet surfaces.
- Temperature contour on the bottom surface of the turbine blade.
- Velocity contour at inlet of the fluid domain.
- Velocity contour at the outlet of the fluid domain.
- Temperature contour along the fluid domain.

And hence the following graphs are being plotted against 15 chosen z-coordinates along the fluid domain in order to analyse their variation for above mentioned mass flow rates.

- Trend of average Nusselt number.
- Trend of average Heat Transfer Coefficient.
- Trend of inner Wall temperature along the fluid domain.

4.1- Temperature and Velocity Contours

- For $m = 0.01$ kg/s

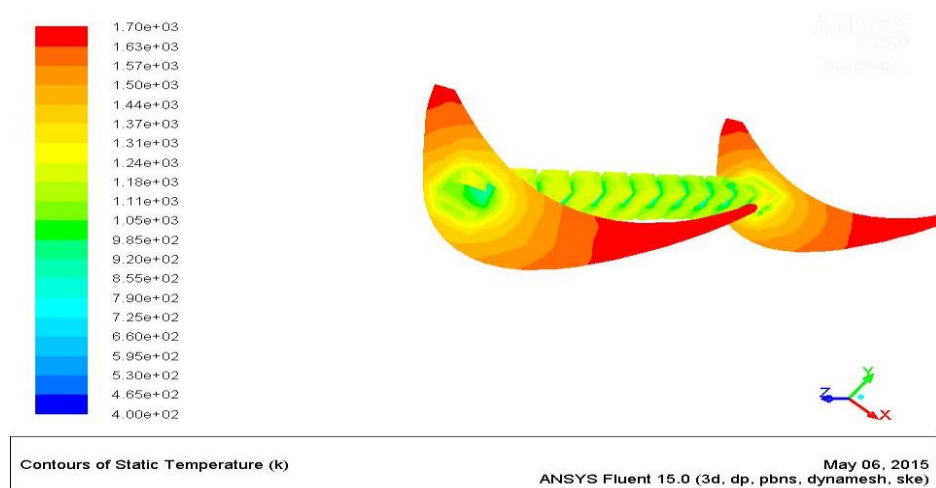


Fig 7 :- Temperature contour on side walls, inlet and outlet

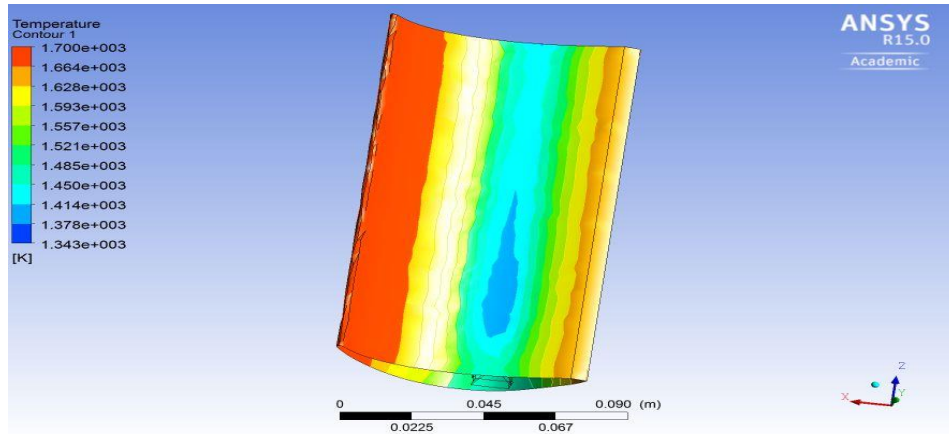


Fig 8:- Temperature contour on the bottom surface of the blade

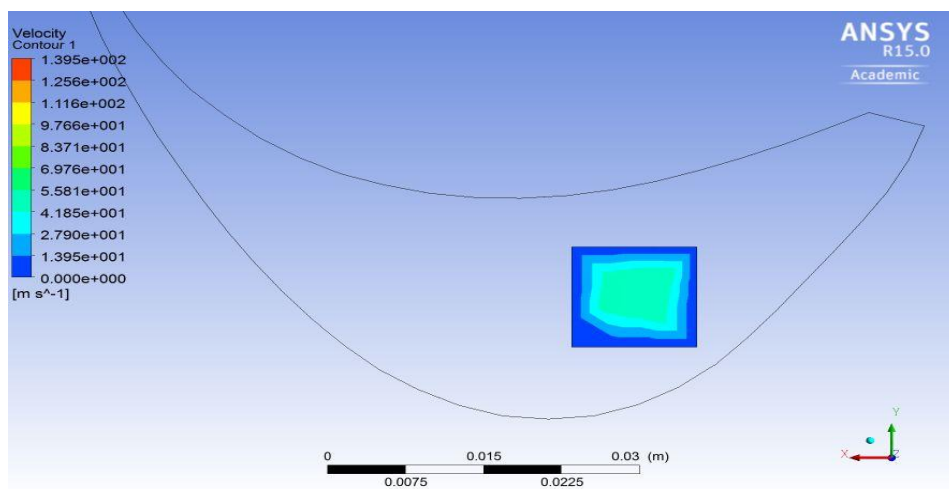


Fig 9 :- Velocity- contour at inlet

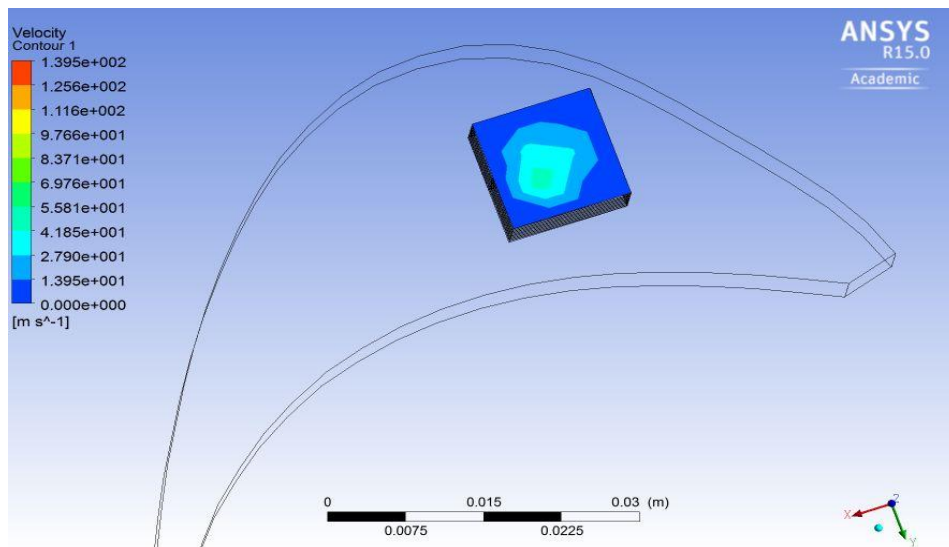


Fig 10 :- Velocity-contour at outlet

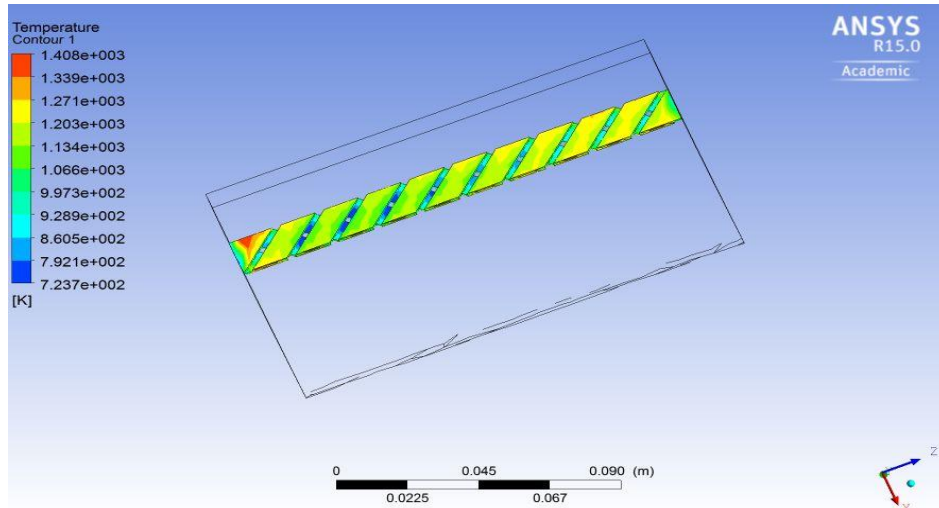


Fig 11 :- Temperature contour along the fluid domain

- For $m = 0.02 \text{ kg/s}$

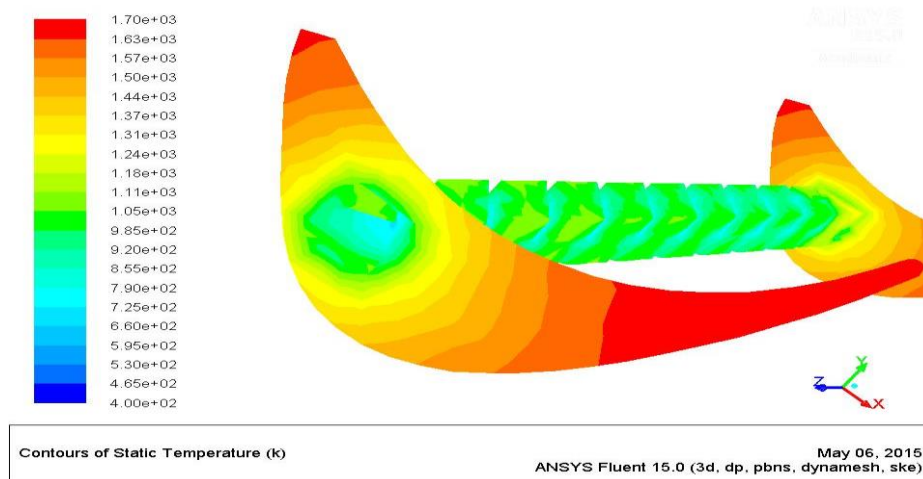


Fig 12 :- Temperature contour on the side walls, inlet and outlet.

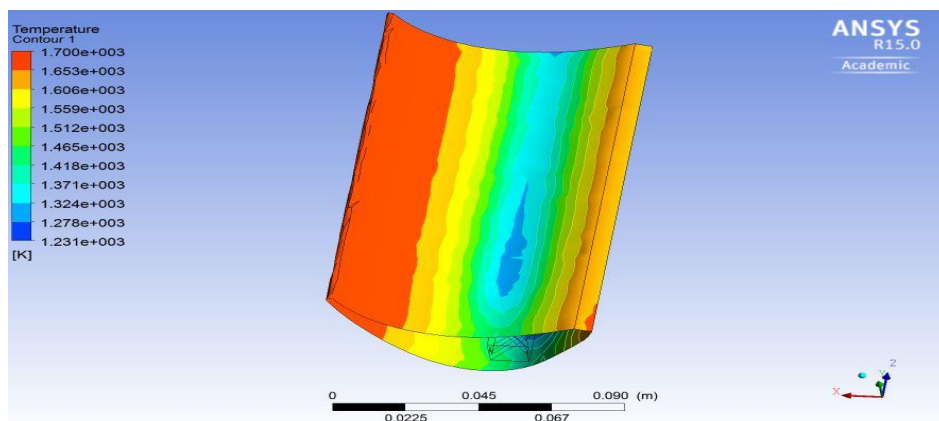


Fig 13 :- Temperature contour on the bottom surface of the blade.

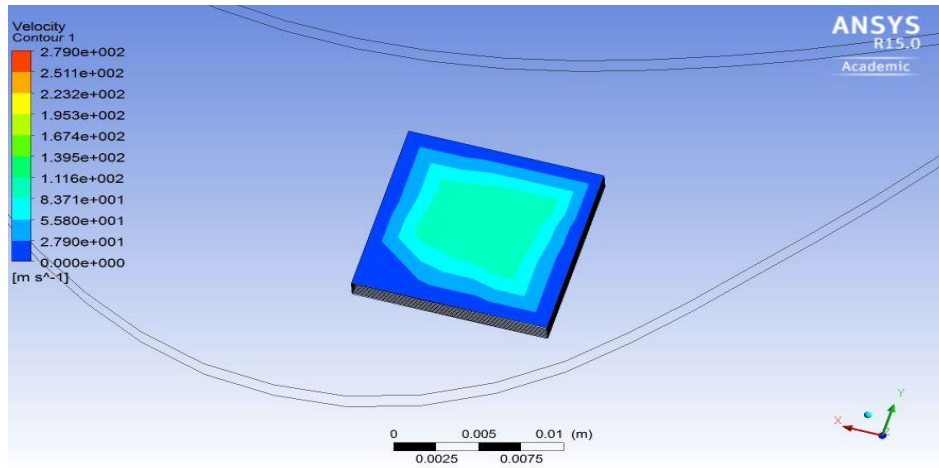


Fig 14 :- Velocity-contour at inlet.

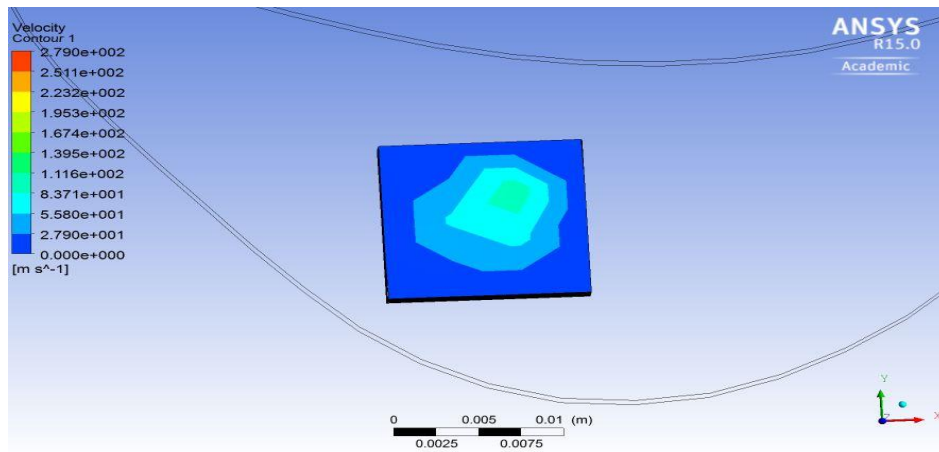


Fig 15 :- Velocity-contour at outlet

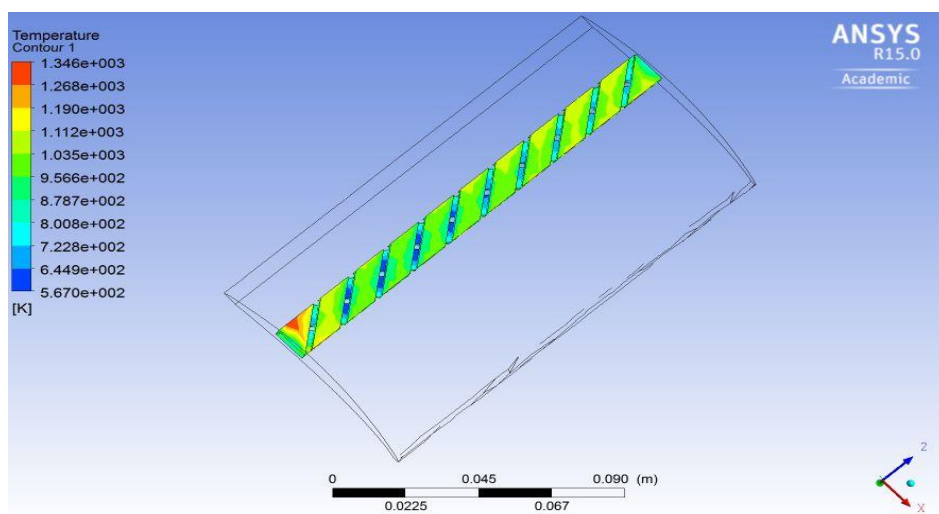


Fig 16 :- Temperature contour along the fluid domain.

- For $m = 0.03 \text{ kg/s}$

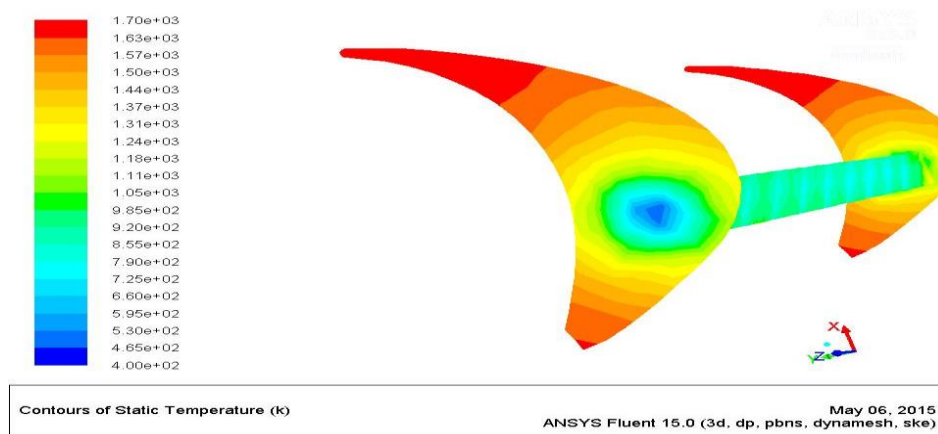


Fig 17 :- Temperature contour on the side walls, inlet and outlet.

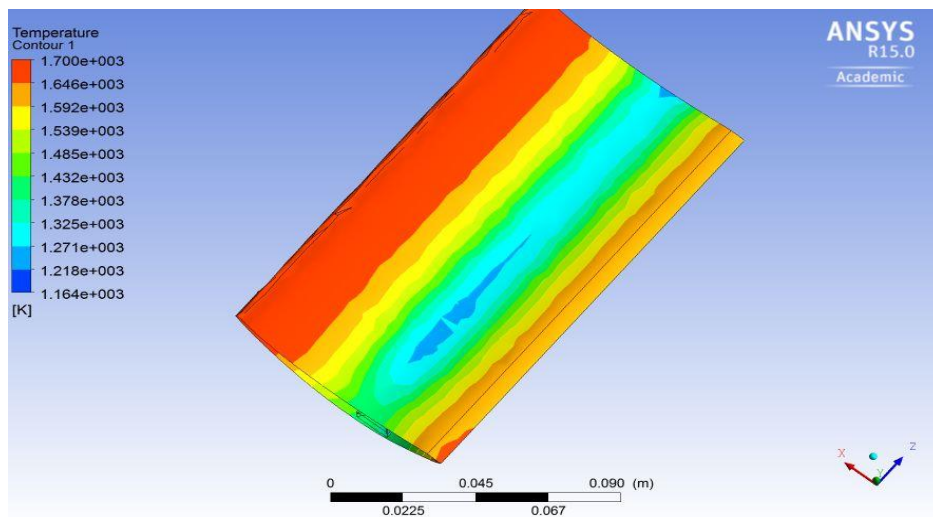


Fig 18 :- Temperature contour on the bottom surface of the blade.

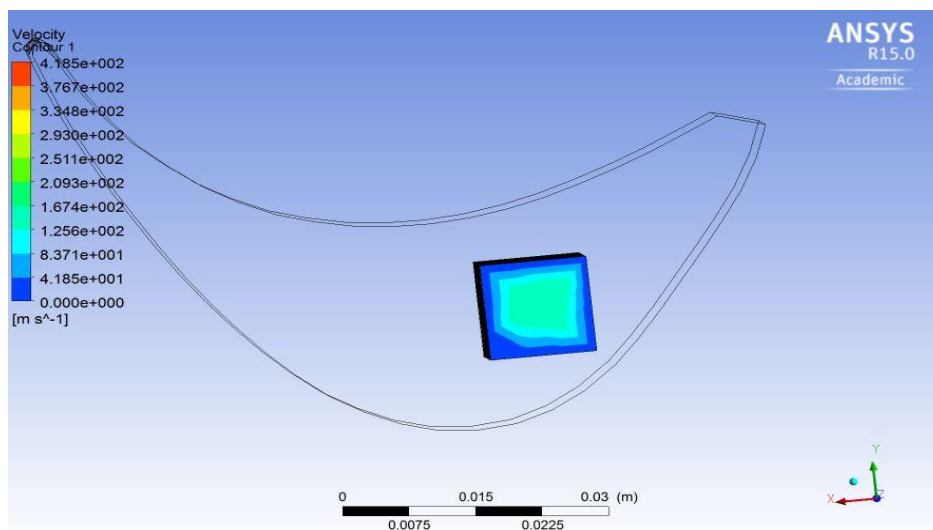


Fig 19 :- Velocity-contour at inlet

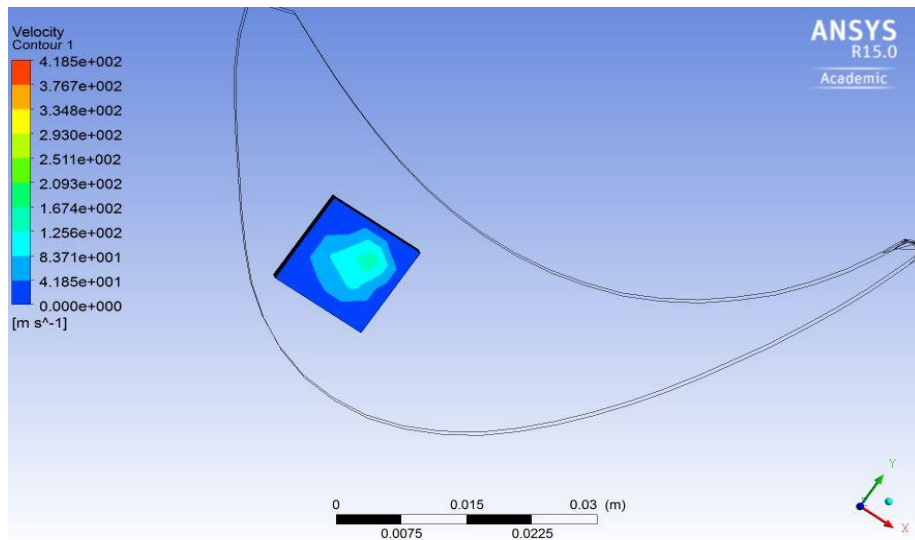


Fig 20 :- Velocity-contour at outlet

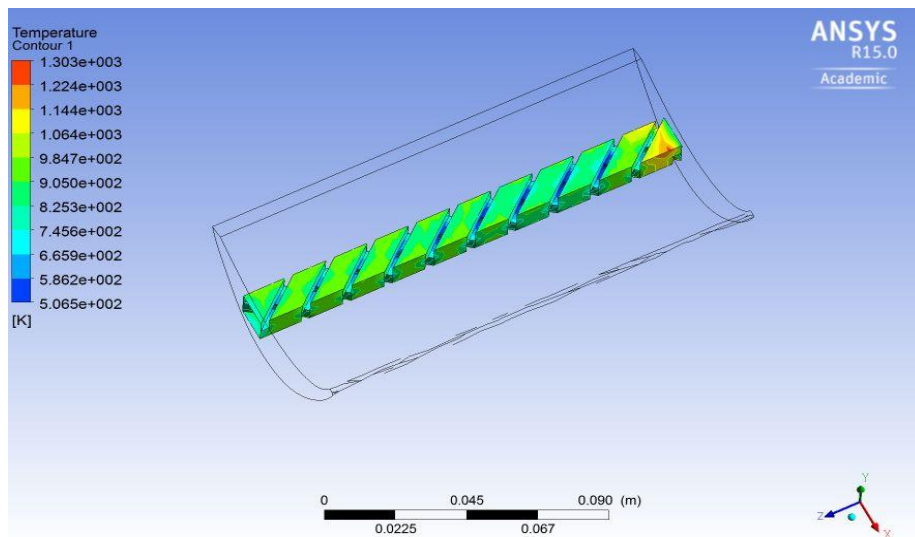


Fig 21 :- Temperature contour along the fluid domain

- For $m = 0.04 \text{ kg/s}$

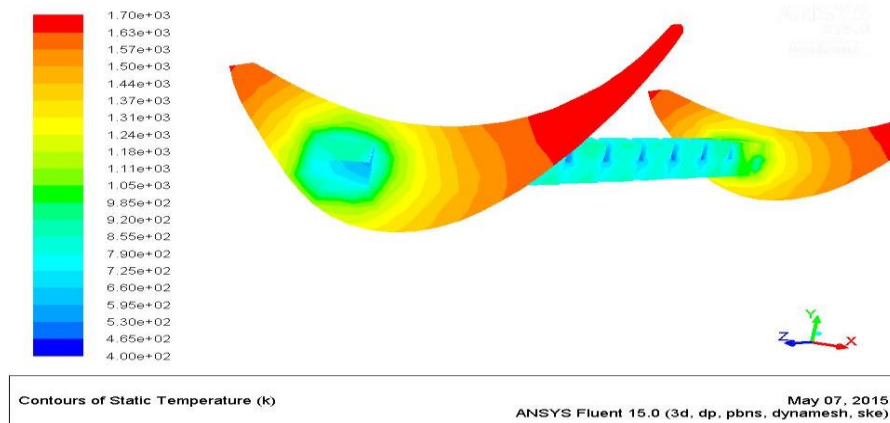


Fig 22 :- Temperature contour on the side walls, inlet and outlet.

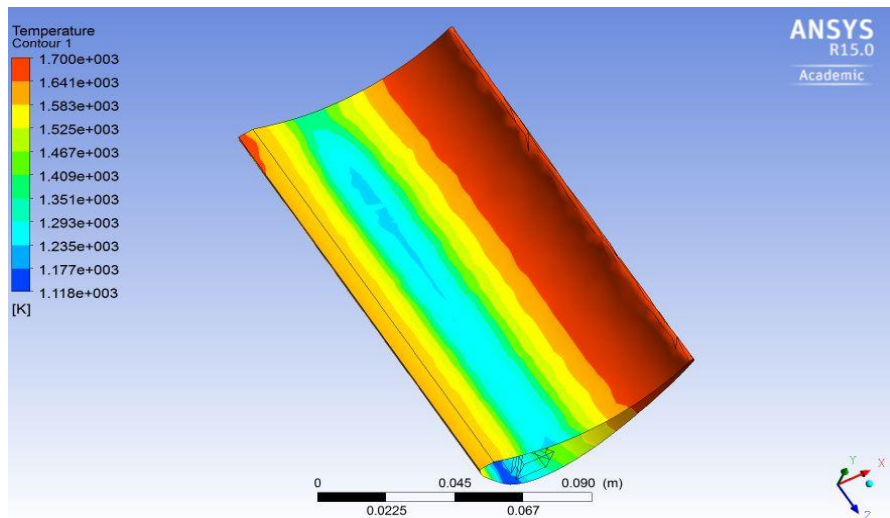


Fig 23 :- Temperature contour on the bottom surface of the turbine blade

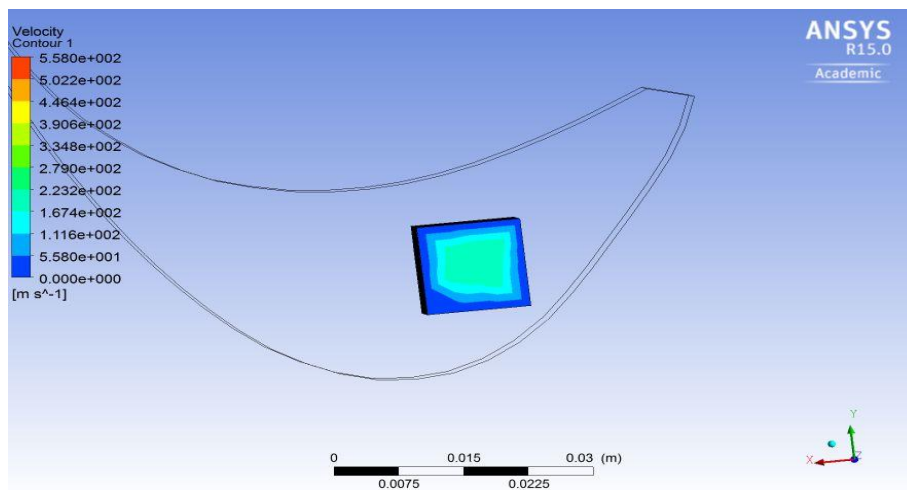


Fig 24 :- Velocity-contour at inlet

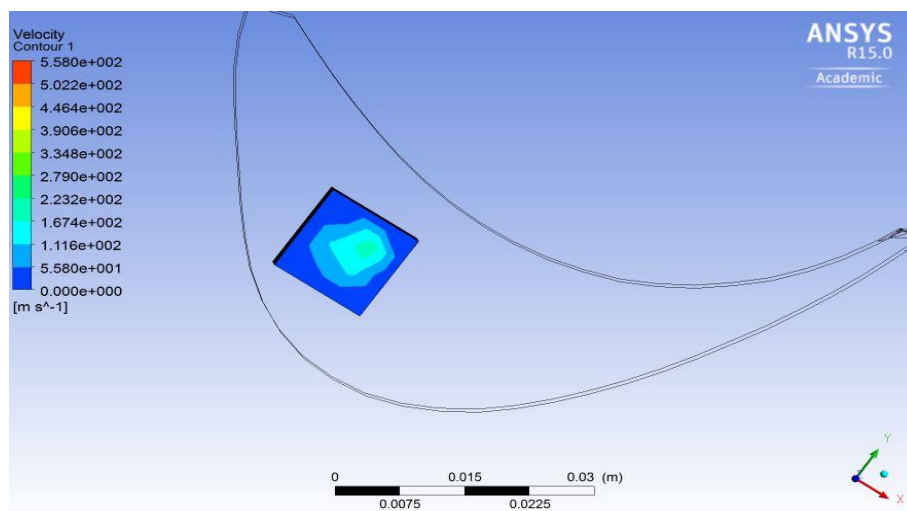


Fig 25 :- Velocity-contour at outlet

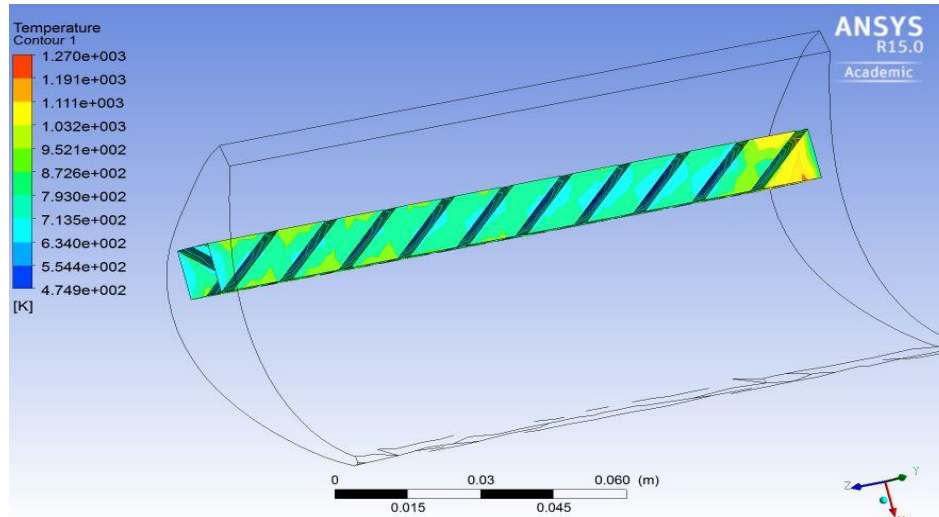


Fig 26 :- Temperature contour along the fluid domain

- For $m = 0.05 \text{ kg/s}$

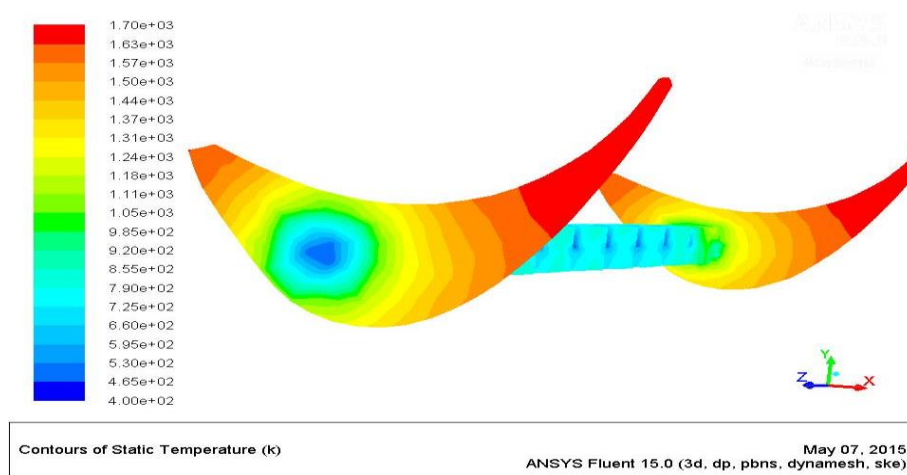


Fig 27 :- Temperature contour on the side walls, inlet and outlet.

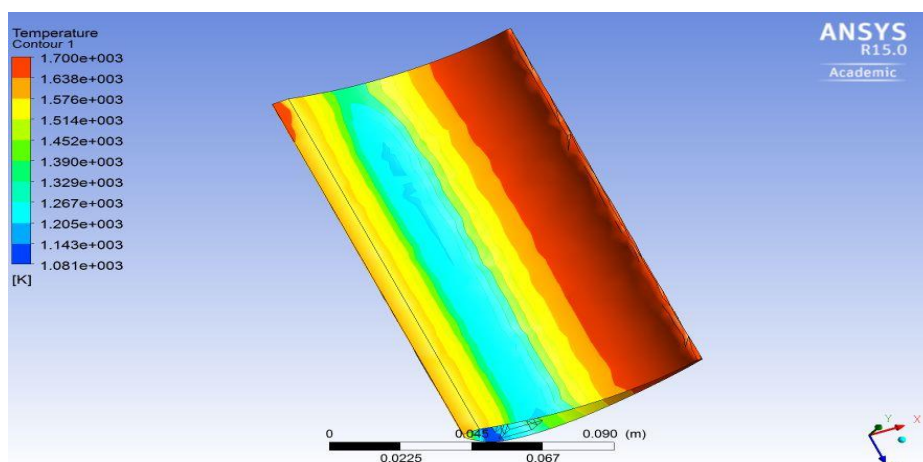


Fig 28 :- Temperature contour on the bottom surface of the blade

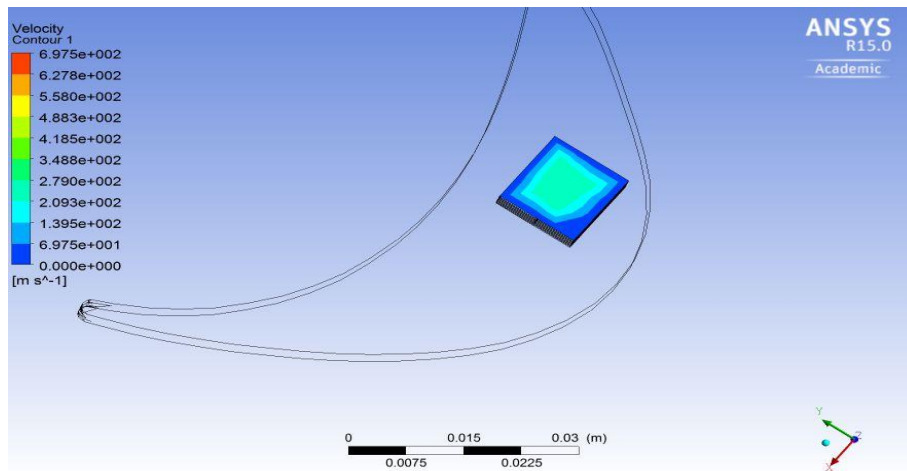


Fig 29 :- Velocity-contour at inlet

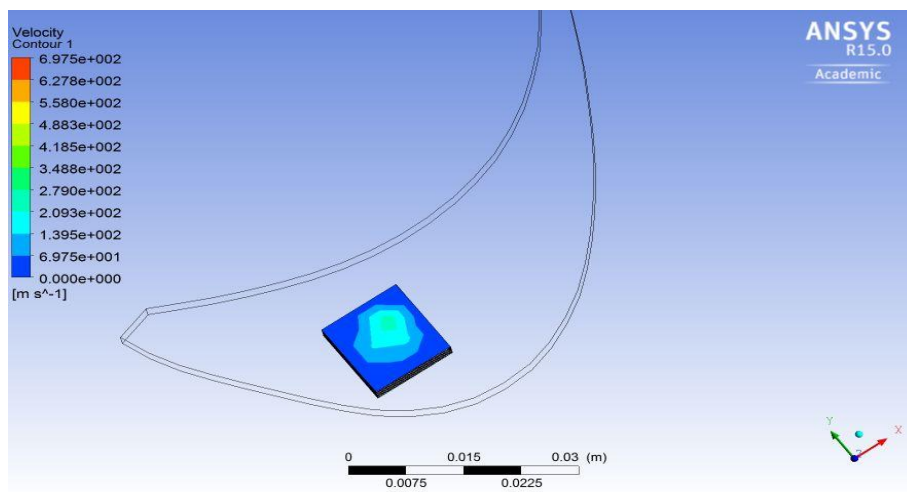


Fig 30 :- Velocity-contour at outlet.

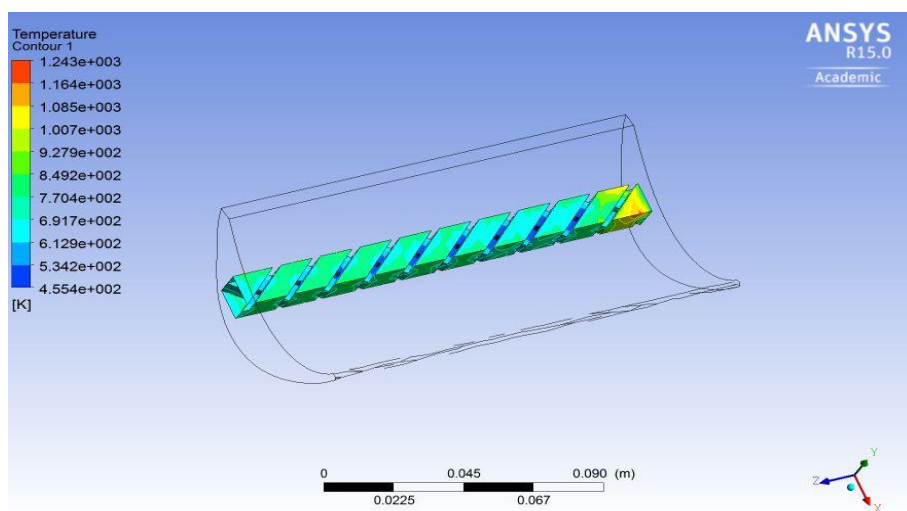
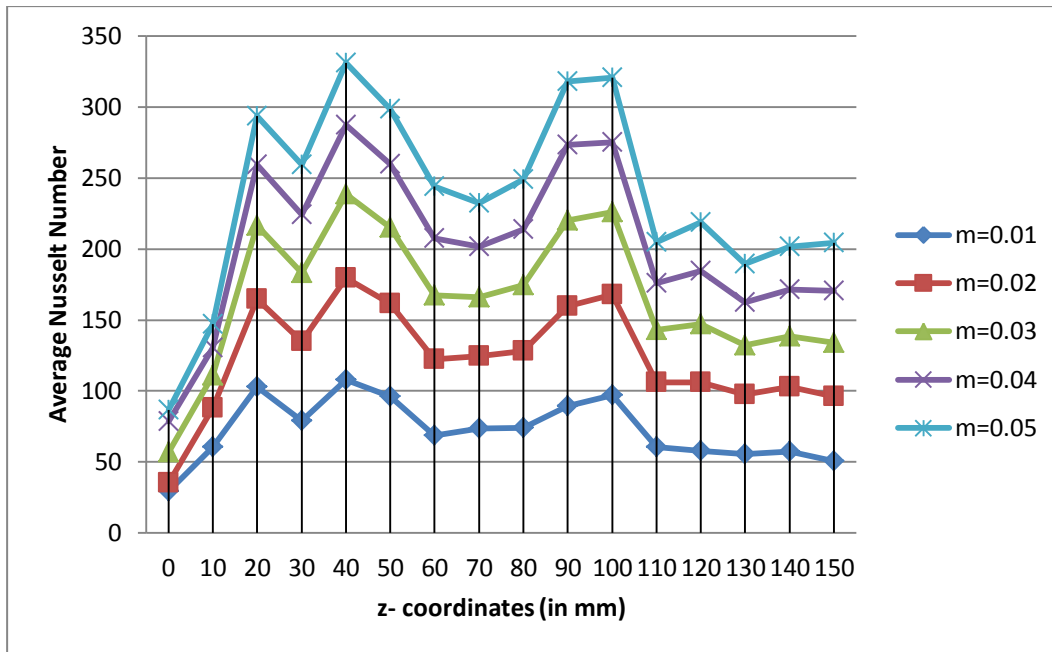
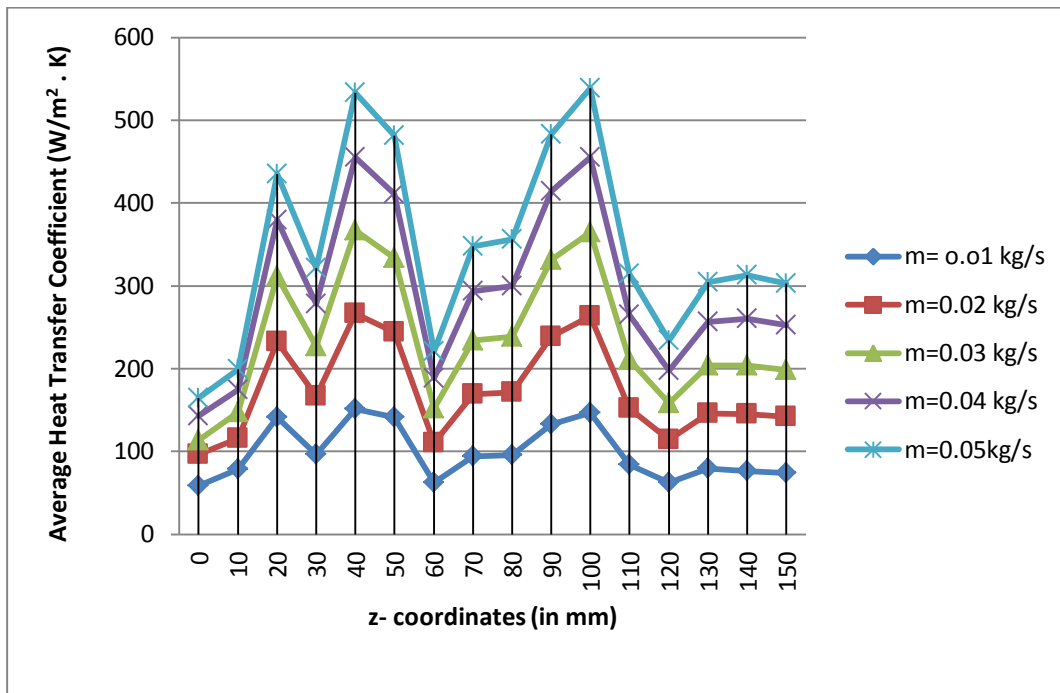


Fig 31 :- Temperature contour along the fluid domain.

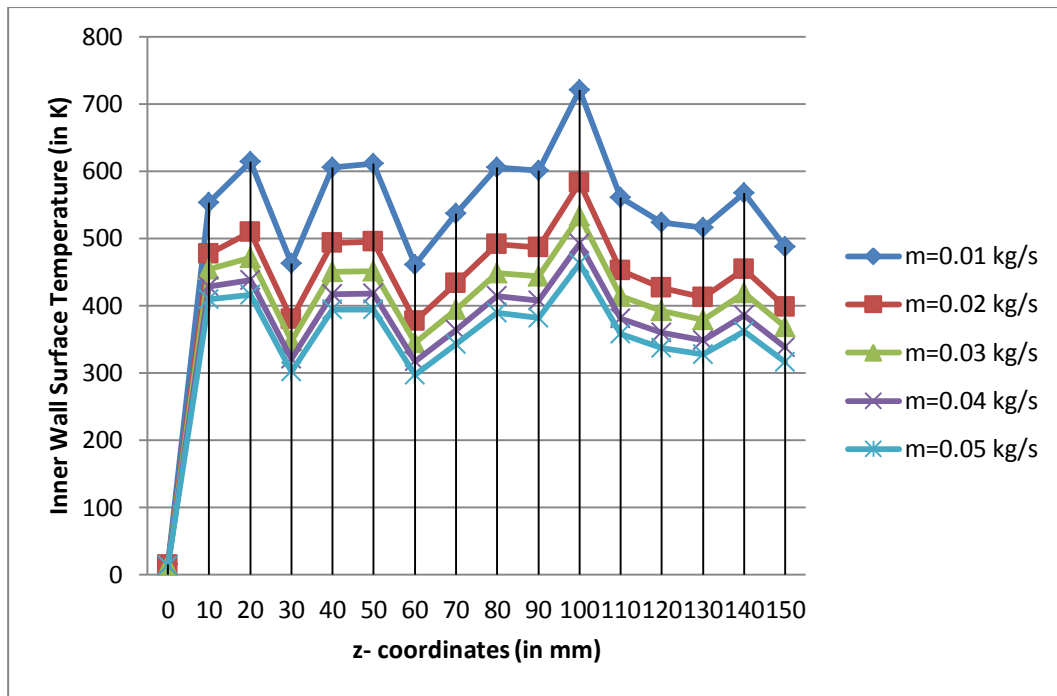
4.2 Graphs and Plots:-



Graph 1 - Plot between Average Nusselt number and chosen z-coordinates.



Graph 2 - Plot between Average Heat Transfer Coefficient and z-coordinates.



Graph 3 - Plot between Inner Wall Surface Temperature and z-coordinates.

5. DISCUSSIONS:-

5.1- Comparison with Smooth Channel

The average Heat Transfer Co-efficient and average Nusselt number at the ribbed matrix channel is found to be more with respect to that of the smooth channel, for a mass flow rate of $m=0.01$ kg/s. Thus, it proves the claim that the ribbed matrix channel enhances the heat transfer between the cooling air and blade, which results in better cooling.

Type of Channel	Average Nusselt Number	Average Heat Transfer Coefficient
Smooth	60.78011	$72.68625 \text{ W/m}^2 \cdot \text{K}$
Ribbed Matrix	72.32946	$96.80817 \text{ W/m}^2 \cdot \text{K}$

5.2- Comparison with Previous results

The simulation was carried on considering all the boundary conditions specified in [2], [15] and the temperature at similar points on the ribbed matrix channel that are derived from this simulation are in the same range. This further validates the result out of this simulation.

5.3- Cases

The cooling effects of air due to increase in the flow rate of mass were increased. This was as expected, as increased mass-flow through the same cross-section increases velocity and results in increase in transfer of heat due to increased turbulence. Thus average heat transfer coefficient and average Nusselt number increased with the increase in mass-flow rate. The temperature of the boundary points decreased with large increment in flow rate. However, increased mass flow through the channel isn't advisable as it lowers the efficiency of the engine at the cost of cooling the blade. The various trends of temperature, Average Nusselt number, heat transfer coefficient are shown in 4.2.

6. CONCLUSION:-

It can be concluded from this work that the claims made by various researchers in the past years as far as the cooling effects of ribbed matrix channels are concerned are true and verified. The ribbed matrix channels offer a significant enhancement in heat transfer. A further hypothesis was tested in this work, by the increase in mass flow rate in search of an optimal flow rate of mass for the flow conditions mentioned. No such optimal mass flow rate could be obtained as the trend was inconclusive. The increase in the heat transfer coefficient and the Nusselt number is monotonous with increasing mass flow rate and thus the trend was inconclusive and vibrant.

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